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DESCRIPTION

IN-LINE FOUR-CYLINDER ENGINE FOR VEHICLE AND VEHICLE PROVIDED WITH THE ENGINE

TECHNICAL FIELD

[0001] The present invention relates to an in-line four-cylinder engine for a vehicle, the engine provided with a primary balancer rotating at the same speed in the opposite direction as and to a crankshaft, and a vehicle provided with the engine.

BACKGROUND ART

[0002] A so-called one-plane 180° type engine, that is, an engine with crankshaft arrangement in which crank pins of first and fourth cylinders are arranged in a same phase while crank pins of second and third cylinders are arranged with a 180° phase difference has been widely used as an engine for a vehicle conventionally.

[0003] In Patent References 1 and 2, proposed is an engine for a vehicle and a vehicle in which a balancer (referred to as a primary balancer) for constant velocity is provided for a crankshaft whose arrangement of each crank pin is 0° for a first cylinder, 90° for a second cylinder, 270° for a third cylinder and 180° for a fourth cylinder, respectively (referred to as a two-plane 90° type).

Patent Reference 1: JP-A-57-69137

Patent Reference 2: JP-A-9-250597

[0004] In a vehicle such as a motorbike provided with an engine using the above-mentioned two-plane 90° type crankshaft, a driver can more strongly feel driving torque from the engine with the whole body of drivers than the case of an engine using the former-mentioned one-plane 180° type crankshaft from a point of view of S/N (SN ratio) of the driving torque, amplitude of vibrations, frequency and such. This allows acceleration feeling to be improved, and thereby, a driving sense to become extremely good. Therefore, it is strongly desired to put the two-plane 90° type engine for a vehicle into practice.

~~DISCLOSURE OF THE INVENTION~~

~~PROBLEMS TO BE SOLVED BY THE INVENTION~~

[0005] An engine in Patent Reference 1 is provided on the premise that the total weight added to a crank web is set by adding 1/2 of the weight W_{rec} of a reciprocating portion to the weight W_{rot} of a rotating portion (refer to a formula (12) in the left column of Page 3). That is to say, $W_{rec}/2$ (additional weight, unbalance weight) is added other than the weight W_{rot} balanced with the weight W_{rot} of the rotating portion such as a large end portion of a connecting rod. Further, it can be seen that calculation is performed on the assumption that the weight of a crank web of each of the cylinders concentrates at the center of the cylinder since the distance from the center of a crankshaft to the center of each of the first and fourth cylinders is set at 2a while the distance to the center of each

of the second and third cylinders is set at 2b (refer to Fig. 1).

[0006] The crank web of each of cylinders is actually divided between the left and the right with respect to the center of the cylinder. Accordingly, in order to assume that the sum of the halves of the crank web concentrates at the center of the cylinder, the additional weight of each of the half crank webs should be 1/4 of the weight of the reciprocating portion. The engine in Patent Reference 1 is provided on the premise that the additional weight to a whole crank web for each cylinder is 1/2 of the weight W_{rec} of the reciprocating portion, namely, the additional weight to a half crank web is 1/4 of the weight of the reciprocating portion.

[0007] In an actual engine design, however, it is often difficult to set the additional weight of a half crank web at 1/4 of W_{rec} since interference with other gears, bearings and the like provided in the vicinity of the crankshaft should be prevented. For the purpose of achieving the same effect as in securing the weight of a half crank web, enlargement of the rotational radius can be considered. In ~~the~~ this case, however, the crankshaft becomes large.

[0008] In the engine proposed in Patent Reference 2, it is assumed that the additional weight added to a crank web is 1/2 of the weight W_{rec} of the reciprocating portion (in Paragraph 0022) and the weight of a crank web for each cylinder

concentrates at the center of the cylinder (in Paragraph 0018), similarly to the engine proposed in Patent Reference 1. Accordingly, the additional weight of each half crank web is assumed to be 1/4 of the weight of the reciprocating portion as a premise. On the basis of the premise, in Patent Reference 2, an isotropic balancer rotating at a constant velocity in an isotropic direction with respect to the crankshaft is provided other than a reverse balancer rotating at a constant velocity in a reverse direction with respect to the crankshaft and a part of the weight $W_{rec}/2$ added to the crankshaft is divided for the isotropic balancer so that the addition weight of the crankshaft is made 1/2 of W_{rec} or less. In ~~the~~ this case, however, a balancer rotating in the isotropic direction should be further added other than a balancer rotating in the reverse direction. This causes unavoidable increase in size of the engine. Summary ...

[0009] The invention has been made for overcoming the above problems, and a first object of the invention is to provide an in-line four-cylinder engine for a vehicle, which is compact and in which acceleration feeling is improved and a driving sense is extremely good.

[0010] Further, a second object of the invention is to provide a vehicle in which S/N of driving torque, amplitude of vibrations, frequency and such of an engine are improved to improve acceleration feeling and to make a driving sense

extremely good.

~~MEANS TO SOLVE THE PROBLEMS~~

[0011] In accordance with the invention, the first object can be achieved by an in-line four-cylinder engine for a vehicle including a crankshaft having crank pins of two cylinders, the crank pins provided on a common first virtual plane in arrangement with a 180° phase difference, and crank pins of the other two cylinders, the crank pins provided on a second virtual plane different by a 90° phase from the first virtual plane in arrangement with a 180° phase difference, the in-line four-cylinder engine for a vehicle comprising: a crankshaft satisfying a formula of $(k_L - 0.25) \cdot (0.25 - k_R) \cong D_R / D_L$, wherein, when a crank web for each of at least two cylinders is divided between a pair of half crank webs faced with respect to a crank pin, k_L, k_R denote balance ratios of the both half crank webs (wherein $k_L \neq 0.25$, $k_R \neq 0.25$) and D_L, D_R denote distance from the center in a longitudinal direction of the crankshaft to the respective half crank webs, the crankshaft being arranged that crank webs for four cylinders be set so that a track of a vector of a primary inertial couple would be formed into a substantially circular shape; and a primary balancer for generating a couple vector offsetting a vector of the first inertia couple.

[0012] The second object can be achieved by a vehicle provided with the in-line four-cylinder engine for a vehicle

according to Claim 1.

EFFECT OF THE INVENTION

[0013] The invention is made on the basis of finding by an inventor that a state that the additional weight W_{add} of half crank webs for at least two cylinders (actually, used are balance ratios k_L and k_R expressed by a ratio of the moment, which is the product of the weight and a rotational radius, and the moment, which is the product of the reciprocation weight and the rotational radius) and distance D_L and D_R from the center of the crank to the half crank webs satisfy a predetermined condition is equivalent to a state that the additional weight W_{add} of a crank web is 1/2 of the weight W_{rec} of the reciprocating portion and the weight of the crank web concentrates at the center of the cylinder.

[0014] That is to say, satisfying the predetermined condition $(k_L - 0.25)/(0.25 - k_R) \cong D_R / D_L$ (wherein $k_L \neq 0.25$, $k_R \cong 0.25$) allows the acceleration feeling to be improved and a driving sense to be made extremely good as well as the additional weight W_{add} of a half crank web to be varied, similar to Patent References 1 and 2.

[0015] Accordingly, a balance ratio, inertia mass and the like of a crank web can be increased in degree of freedom in design since a half crank web for each cylinder can be changed in shape, so that the engine can be made compact. Moreover, in accordance with the second aspect of the invention, it is

possible to achieve a vehicle in which acceleration feeling is improved and a driving sense is extremely good.

BRIEF DESCRIPTION OF THE DRAWINGS

[0016] Fig. 1 is a side view of a schematic structure of an in-line four-cylinder engine for a vehicle in accordance with the invention.

Fig. 2 is a plan view of a crankshaft and a couple balancer.

Fig. 3 is a perspective view of a crankshaft and a couple balancer.

Fig. 4 is a sectional view for illustrating an operation of a crankshaft and a balancer.

Fig. 5 is a perspective view of a simplified structure of a crankshaft.

Fig. 6 illustrates crank angles.

Fig. 7 is a graph showing variations in primary inertia force for every cylinder.

Fig. 8 illustrates a simplified structure of a reciprocating portion.

Fig. 9 is a perspective view for illustrating a compound couple vector (T).

Fig. 10 illustrates compound vectors (a to d) of primary inertia force for every cylinder and a compound couple vector (e) of a whole engine.

Fig. 11 is a perspective view showing balance ratios of

crank webs.

Fig. 12 illustrates forms of a crankshaft.

Fig. 13 shows balance ratios and distance D.

Fig. 14 is a side view of a motorbike provided with an engine.

DESCRIPTION OF SIGNS AND NUMERALS

[0017] 1, 50: engine

3, 56: crankshaft

6, 58: couple balancer

12: first balance weight

13: second balance weight

21 to 24: crank pin

21a, 21b, 22a, 22b, 23a, 23b, 24a, 24b: half crank web

p_1: first virtual plane

p_2: second virtual plane

BEST MODE FOR CARRYING OUT THE INVENTION

[0018] A balance ratio k for one crank web formed from a pair of half crank webs is defined by the following formula (1):

$$k=(M-W_{rot}\cdot r/2)/(W_{rec}\cdot r) \quad \cdots(1)$$

[0019] wherein M denotes total unbalance quantity of a crank web (whose unit is the moment g·mm), r denotes the rotational radius of W_{rot} and W_{rec}. r is 1/2 of a rotational circle of a crank pin and also 1/2 of a stroke of the reciprocating portion. W_{rot} denotes mass for a rotating

portion and W_{rec} denotes mass for a reciprocating portion (whose unit is g).

[0020] $(M/r - W_{rot}/2) = W_{add}$ is usually set at a half of the mass for a reciprocating portion W_{rec}. Accordingly, $k = 1/2$ (50%) in the above case. In the invention, a crank web for a cylinder is divided between the left and the right to form half crank webs, and then, the balance ratios k_L and k_R of the crank webs are separately set. Thus, in the above case, k_L and $k_R = (M - W_{rot} \cdot r/4)/(W_{rec} \cdot r)$ when M is assumed to be the unbalance quantity of a half crank web in the formula (1). The predetermined requirement is determined in view of the distance D_L and D_R of the respective crank webs from the center of the crankshaft under the above condition. The meaning of the requirement will be described later.

[0021] In the invention, it is clear that the crankshaft can be reduced in weight when $(k_L + k_R)$ is made smaller than 0.5 (50%). This is especially suitable for an engine for a vehicle (Claim 2). Contrary to the above, making $(k_L + k_R)$ larger than 0.5 (50%) (Claim 3) means that the half web on a side of the center of the crankshaft is increased in weight as described later. In this case, however, degree of freedom in design is increased in preventing interference with other members or in providing other gears or the like on the crankshaft. Further, degree of freedom in determining mass of a half web for each cylinder is increased and the weight

concentrates on the center side of the crankshaft. This is suitable for keeping down torsional vibrations of the crankshaft, so that decrease in weight of the crankshaft can be achieved.

[0022] In the invention, it is possible to arrange a requirement in Claim 1 to be satisfied for two cylinder and the balance ratios k_L and k_R of the left and right half crank webs to be respectively set at 0.25 and 0.25 for the other two cylinders, similarly to the case of a conventional engine (in Claim 4). As described above, the invention can be only applied to arbitrary two cylinders for reasons of layout of an engine. An effect of an increase in degree of freedom in design can be achieved in the above case.

[0023] As for crank pins of a crankshaft, a various kind of arrangement can be considered. Fig. 12 shows forms of the arrangement. In the mode for carrying out the invention, it is defined that an X-Z plane is a first virtual plane P1 and an Y-Z plane is a second virtual plane P2 wherein X, Y and Z axes cross at right angles, as shown in Fig. 12(A). In Fig. 12(A), first to fourth cylinders are provided in this order from the left end and crank pins of the first and fourth cylinders are located on the first virtual plane P1 while crank pins of the second and third cylinders are located on the second plane P2. Such a form is referred to as STD arrangement hereinafter (Claim 5). In the crankshaft in the STD

arrangement, primary inertia force, secondary inertia force and a secondary inertia couple can be theoretically made 0 because of symmetry of crank webs. It is also possible in theory to form a vector diagram of a primary inertia couple into an exact circle and completely remove it by means of a primary balancer.

[0024] In the following description, used are words "offset", "remove", "eliminate", "made to be 0" and the like for the inertia force, the couple and such. They mean reduction to the extent that there is practically no problem and do not necessarily mean only reduction to 0. The number of the primary balancer shaft to be provided is preferably one in order to make an engine compact. It is also possible, however, to divide the balancer shaft into two or more in the invention.

[0025] In Fig. 12(B), crank pins of the first and third cylinders are provided on the first virtual plane while crank pins of the second and fourth cylinders are provided on the second virtual plane. Such a form will be referred to as 90-I arrangement hereinafter (Claim 6). In Fig. 12(C), crank pins of the first and second cylinders are provided on the first virtual plane while crank pins of the third and fourth cylinders are provided on the second virtual plane. Such a form will be referred to as 90-J arrangement hereinafter (Claim 7).

[0026] In the 90-I and 90-J arrangement, at least one of

the primary inertia force and the secondary inertia couple remains a little or the vector diagram of the primary inertia couple is formed into an ellipse close to a circle instead of an exact circle. Vibrations, however, can be reduced to the extent that there is no problem in practical use.

[0027] In the case of the STD arrangement, it is preferable to provide the first and fourth cylinders to be symmetrical (mirror arrangement) in balance ratio k_L and k_R and distance D_L and D_R with respect to the center of the crank and to provide the second and third cylinders to be symmetrical (mirror arrangement) with respect to the center of the crank (Claim 8). In this case, adding the primary balancer theoretically allows the primary and secondary inertia force and couples to be completely removed, as described above. A few vibrations, however, may be left by asymmetrical configuration for the purpose of reduction in weight or improving in drive feeling in the invention.

[0028] In the case of the 90-I and 90-J arrangement, the first and fourth cylinders and the second and third cylinders may be provided so as to be symmetrical in distance D_L and D_R, respectively, while arbitrarily combined two cylinders may be provided so as to be symmetrical (mirror arrangement) in balance ratio k_L and k_R (Claim 9). In this case, one or both of the primary inertia force and the secondary inertia couple remains a little or the vector diagram of the primary inertia

couple is formed into an ellipse. It is possible, however, to leave vibrations to the extent that there is practically no problem in use to achieve reduction in weight or improvement in drive feeling.

[0029] The balance weight of the primary balancer can be provided at a place opposite to the crank pins of the second and third cylinders (between the half crank webs) or a place opposite to the crank pins of the first and fourth cylinders (between the half crank webs) (Claim 10). In this case, it is possible to provide the balancer weight closely to the crankshaft to make an engine compact.

EMBODIMENT 1

[0030] An embodiment of the in-line four-cylinder engine in accordance with the invention will be described in detail hereinafter on the basis of Figs. 1 to 11.

Fig. 1 is a side view of a schematic structure of an in-line four-cylinder engine in accordance with the invention. Fig. 2 is a plan view of a crankshaft and a couple balancer. Fig. 4 is a sectional view for illustrating an operation of a crankshaft and a balancer. Fig. 5 is a simplified perspective view of a crankshaft. Fig. 6 illustrates crank angles. Fig. 7 is a graph showing variations in primary inertia force. Fig. 8 illustrates a simplified structure of a reciprocating portion. Fig. 9 is a perspective view for illustrating a compound couple vector (T). Fig. 10

illustrates compound vectors (a to d) of primary inertia force for every cylinder and a compound couple vector T(e) of a whole engine. Fig. 11 is a perspective view showing balance ratios of crank webs.

[0031] In the above drawings, Numeral 1 denotes a water-cooled four-cycle in-line four-cylinder engine in Embodiment 1. The engine 1 is for a motorbike. In the engine 1, a later-mentioned crankshaft 3 is rotatably held in a crankcase 2 formed so as to be capable of division in a vertical direction. An axial line of the crankshaft 3 of the engine 1 is in parallel to a width direction of a vehicle. The engine 1 is mounted to a vehicle body frame (not shown) so that a left portion in Fig. 1 would be located on a front side of a vehicle body.

[0032] The crankcase 2 is formed from an upper crankcase main body 4 and a lower crankcase main body 5 between which the crankshaft 3 and a later-mentioned couple balancer 6 are held so as to be able to rotate freely. At an upper end portion of the upper crankcase main body 4, mounted is a cylinder body 7. The lower crankcase main body 5 is provided with a main shaft 8 and a driving shaft 9 so that they can freely rotate. At a lower end portion of the lower crankcase main body 5, mounted is an oil pan 5a. The cylinder body 7 is formed so that cylinder holes 7a for four cylinders would be in line in the width direction of a vehicle and is provided at the upper

end thereof with a cylinder head (not shown). The cylinder hole 7a is formed so that an axial line C would slant rearward. In Embodiment 1, the four cylinders are referred to as a first cylinder, a second cylinder, a third cylinder and a fourth cylinder in this order from the left end of the vehicle body.

[0033] The main shaft 8 and the driving shaft 9 are connected to each other via a transmission (not shown) having a conventionally well-known structure. The main shaft 8 is provided at the right end portion of the vehicle with a clutch (not shown) mounted on the shaft. The main shaft 8 is connected by means of gears to the couple balancer 6 through the clutch and large reduction gear (not shown). The driving shaft 9 is provided at the left end portion of the vehicle with a sprocket (not shown) and is connected to a rear wheel (not shown) through a chain 10 provided around the sprocket for driving a rear wheel.

[0034] The couple balancer 6 is for removing the primary couple generated in accordance with rotation of the later-mentioned crankshaft 3. The couple balancer 6 comprises a balancer shaft 11 rotatably held in the crankcase 2 by means of bearings 2a to 2c and first and second balance weights 12 and 13 formed into one body with the balancer shaft 11, as shown in Fig. 2. The balancer shaft 11 is provided with a driven gear 14 between the both balance weights 12 and 13 and with small reduction gear 15 at the right end portion of the vehicle.

[0035] The driven gear 14 is formed so that the number of rotation thereof is same as that of an output gear 16 of the crankshaft 3. The driven gear 14 is engaged with the output gear 16 while the small reduction gear 15 is engaged with the large reduction gear provided on the clutch side. That is to say, in the engine 1, rotation of the crankshaft 3 is transmitted to the main shaft 8 through the couple balancer 6 and the clutch, further transmitted from the main shaft to the driving shaft through the transmission, and then, transmitted from the driving shaft to a rear wheel via a chain for driving a rear wheel. A direction of rotation of the crankshaft 3 is a clockwise direction in Fig. 1 in Embodiment 1.

[0036] The crankshaft 3 comprises crank pins 21 to 24 for every cylinder, first and second crank webs 21a, 21b, 22a, 22b, 23a, 23b, 24a and 24b and journal portion 25, as shown in Figs. 1 to 3 and Fig. 5. The crankshaft 3 is provided at the center part in the axial direction of the crankshaft 3 with the output gear 16 and a crank angle sensor wheel 26. In Figs. 1 and 4, 27 denotes a connecting rod, 28 denotes a piston and 28a denotes a pin piston.

[0037] Locations (crank angles) in a rotational direction of the respective crank pins 21 to 24 are set so that the crank angle of the crank pin 22 for the second cylinder would be 270° with respect to the crank pin 21 for the first cylinder, the

crank angle of the crank pin 23 for the third cylinder would be 90° with respect to the crank pin 21 for the first cylinder and the crank angle of the crank pin 24 for the fourth cylinder would be 180° with respect to the crank pin 21 for the first cylinder, as shown in Fig. 6. That is to say, the crankshaft 3 is of the two-plane type (the STD arrangement) in which the crank pin 21 for the first cylinder and the crank pin 24 for the fourth cylinder are located on the first virtual plane while the crank pin 22 for the second cylinder and the crank pin 23 for the third cylinder are located on the second plane and in which the both planes cross at right angles. Ignition of the engine 1 is carried out in the order of the first cylinder, the third cylinder, the second cylinder and the fourth cylinder.

[0038] The crank web is provided on its opposite side with respect to the core of the crankshaft 3 with a counter weight portion. Mass of the counter weight portion is set so as to be able to reduce vibration force of the engine 1 in cooperation with the couple balancer 6. The mass is the sum of the weight W_{rot} for balancing with the weight W_{rot} of a rotating portion such as a large end portion of a connecting rod and the additional weight W_{add} for balancing with a reciprocating portion. It has been found that setting the additional weight W_{add} at about 50% of the mass of the reciprocating portion {(the piston 28, the piston pin 28a and a small end part 27a of the

connecting rod 27 (refer to Fig. 1)} of the engine 1 (the balance ratio is about 50%) allows the size of the couple operating in a rotation to be appropriate.

[0039] An appropriate size of couple in the above context means a couple in the size that the couple balancer 6, which can be mounted to the engine 1, can remove. In the two-plane type crankshaft 3 in Embodiment 1, the primary and secondary inertia force and the secondary couple are eliminated as described later. Accordingly, a track of movement of a couple compound vector of the remaining primary couple is a circle about the core of the crankshaft when the crank web is formed so that the balance ratio would be about 50%. That is to say, in the engine 1 provided with the crankshaft 3, providing the couple balancer 6 generating a couple, which is to be a couple vector for offsetting the primary couple compound vector, allows the primary couple to be also eliminated. Now, described hereinafter will be the reasons why the primary and secondary inertia force and the secondary couple are eliminated and a structure of the couple balancer 6 for eliminating the primary couple.

[0040] (1) Reason why the primary inertia force is eliminated

The primary inertia force operates upon the crankshaft 3 so as to correspond to reciprocation of the piston 28, as shown in Fig. 7. In Fig. 7, shown are variations of the primary

inertia force due to reciprocation mass of the respective cylinders, the variations being divided between an X-axis component and a Y-axis component. As for the X-axis component, which is in the same direction as that of the piston 28, the inertia force due to the reciprocation mass of the both cylinders is offset by each other since the piston 28 of the first cylinder is at an upper dead point (Point A in Fig. 7) while the piston 28 of the fourth cylinder is at a lower dead point (Point B in Fig. 7), as show in Fig. 7. Similarly, the primary inertia force due to the reciprocation mass of the second and third cylinders is also offset by each other. Accordingly, the primary inertia force of the crankshaft 3 becomes 0 in theory.

[0041] (2) Reason why the secondary inertia force is eliminated

The second inertial force F1 to F4 due to the reciprocation mass of the respective cylinders is expressed by the following formulas (1) to (4) when the size and mass of the respective portions are defined as shown in Fig. 8. In Fig. 8, m_r denotes mass of a reciprocating portion (g), L denotes the length of a connecting rod (mm), r denotes a piston stroke /2 (mm) and ω denotes 2 πN/60 (rad). In the formulas (1) to (4), λ denotes the length of a connecting rod /r.

[0042] [Formula 1]

$$F1 = m_r r\omega^2 \times \left(\frac{1}{\lambda}\right) \times \cos 2\theta \qquad \cdots(1)$$

$$F2 = m_r r\omega^2 \times \left(\frac{1}{\lambda}\right) \times \cos\{2(\theta + 3\pi/2)\} \qquad \cdots(2)$$

$$F3 = m_r r\omega^2 \times \left(\frac{1}{\lambda}\right) \times \cos\{2(\theta + \pi/2)\} \qquad \cdots(3)$$

$$F4 = m_r r\omega^2 \times \left(\frac{1}{\lambda}\right) \times \cos\{2(\theta + \pi)\} \qquad \cdots(4)$$

[0043] The sum F(2) of F1 to F4 becomes 0 as shown in the following description. Accordingly, the secondary inertia force of the crankshaft 3 theoretically becomes 0.

[0044] [Formula 2]

$$F(2) = m_r r\omega^2 \times \frac{1}{\lambda}\left\{\cos 2\theta + \cos 2\left(\theta + \frac{3}{2}\pi\right) + \cos 2\left(\theta + \frac{\pi}{2}\right) + \cos 2(\theta + \pi)\right\}$$

$$F(2) = m_r r\omega^2 \times \frac{1}{\lambda}\{\cos 2\theta + \cos(2\theta + 3\pi) + \cos(2\theta + \pi) + \cos(2\theta + 2\pi)\}$$

wherein $\cos(\pi \pm \theta) = -\cos\theta$

$\cos(2n\pi + \theta) = \cos\theta$, and therefore,

$$F(2) = m_r r\omega^2 \times \frac{1}{\lambda}\{\cos 2\theta - \cos 2\theta - \cos 2\theta + \cos 2\theta\}$$

$$F(2) = 0$$

[0045] (3) Reason why the secondary couple is eliminated

The secondary couple is the sum of the moment around the Y axis, which is generated in accordance with an operation of the secondary inertia force F1 to F4 upon the crankshaft 3, as shown in Fig. 5. The Y axis is an axis crossing at right angles with the X axis, which is parallel to an axial line of a cylinder, and extending in a direction orthogonal to the Z axis, which is parallel to an axial line of the crankshaft 3. Fig. 5 shows a condition that the piston 28 of the first cylinder

is located at the upper dead point.

[0046] The secondary inertia force F1 to F4 is repeatedly generated so that a cycle thereof would be 180° in crank angle. Accordingly, a direction in which the secondary inertia force operates is same between the first and fourth cylinders whose crank angles are different by 180° from each other. The second cylinder whose crank angle is different by 90° to the front side in a direction of the rotation with respect to the first cylinder and the third cylinder whose crank angle is different by 90° to the rear side in a direction of the rotation are opposite to the first and fourth cylinders in direction upon which the secondary inertia force operates. Therefore, the secondary couple FL formed by the sum of the moment is 0 as shown below when d1 to d4 denote the distance from the Y axis to the respective cylinders, d1 = d4 and d2 = d3.

$$FL = F1 \times d1 - F2 \times d2 + F3 \times d3 - F4 \times d4 = 0$$

Thus, the secondary couple of the crankshaft 3 is 0.

[0047] (4) Structure of the couple balancer 6 for eliminating remaining primary couple

The primary couple operating upon the crankshaft 3 can be expressed by the sum of vectors of the primary inertia force operating upon the respective crank pins 21 to 24. This will be described in detail with respect to Fig. 9. Fig. 9 shows a condition that the piston 28 of the first cylinder is located at the upper dead point, namely, a condition that the crank

the inertia force of the rotating portion (operating so as to be directed to the rear of the vehicle) operates in the second cylinder, so that the compound vector of the primary inertia force is directed rearward in parallel to the Y axis (rightward in Fig. 9). The size of the compound vector of the second cylinder varies as shown in Fig. 10(b). A direction of rotation of the compound vector is counterclockwise, inversely to the direction of rotation of the crankshaft.

[0050] In the third cylinder, the inertia force of the reciprocating portion is substantially 0 since the piston 28 is located at a substantially middle point on the way from the upper dead point to the lower dead point, so that only the inertia force of the rotating portion operates. Accordingly, the compound vector of the primary inertia force of the third cylinder is directed forward in parallel to the Y axis (leftward in Fig. 9). The size of the compound vector of the third cylinder varies as shown in Fig. 10(c). A direction of rotation of the compound vector is counterclockwise, inversely to the direction of rotation of the crankshaft.

[0051] In the fourth cylinder, the inertia force of the reciprocating portion is directed downward since the piston 28 is located at the lower dead point, so that the inertia force of the rotating portion, which is about a half of that of the reciprocating portion, is directed upward. Accordingly, the compound vector of the primary inertia force of the fourth

cylinder is directed downward in parallel to the X axis. The size of the compound vector of the fourth cylinder varies as shown in Fig. 10(d). A direction of rotation of the compound vector is counterclockwise, inversely to the direction of rotation of the crankshaft.

[0052] The compound vector of the first cylinder and the compound vector of the fourth cylinder are substantially same in size and opposite by 180° in direction. Accordingly, the primary inertia force operating upon the crank pins 21 and 24 of the both cylinders forms a couple. The couple is referred to as a first couple hereinafter. The first couple is a couple passing through the center line in the axial direction of the crankshaft 3, the couple being for rotating the crankshaft 3 about a virtual axial line YC parallel to the Y axis, in Fig. 9. A vector of the first couple is in a direction that a right screw goes when the couple is in the same direction as the rotational direction in fastening the screw. That is to say, the vector of the first couple is directed forward along the virtual axial line YC. In Fig. 9, a plane in which the virtual axial line YC is located and which is orthogonal to the axial line of the crankshaft 3 is shown as a virtual plane I at the left end of the drawing for the purpose of easy understanding. In the virtual plane I, A denotes the vector of the first couple.

[0053] On the other hand, the compound vector of the second cylinder and the compound vector of the third cylinder

are substantially same in size and opposite by 180° in direction. Accordingly, the primary inertia force operating upon the crank pins of the both cylinders causes a second couple. A vector of the second couple is in a direction that a right screw goes when the couple is in the same direction as the rotational direction in fastening the screw. Therefore, the vector of the second couple passes through the center line in the axial direction of the crankshaft 3 and is directed upward along a virtual axial line XC parallel to the X axis, in Fig. 9. In the virtual plane I, B denotes the vector of the second couple.

[0054] Compounding the vectors A and B allows direction and size of the total couple operating upon the engine 1 to be obtained. A compound couple vector T of the vectors A and B is directed to an obliquely upper front side. The size of the compound couple vector T varies as shown in Fig. 10(e). A direction of rotation of the compound couple vector T is also counterclockwise, inversely to the direction of rotation of the crankshaft.

[0055] The compound couple vector T indicates whole primary couple of the engine 1. Accordingly, generating a couple of a vector, which has point symmetry with the compound couple vector T, (shown by a broken line t in Fig. 9) by means of the couple balancer 6 allows the whole primary couple generated in the engine 1 to be eliminated. A vector t having point symmetry with the compound couple vector T will be

referred to as a balancer vector hereinafter. In order to balance the couple compound vector T and the balancer vector t, most effective is to make a track of rotation of the compound couple vector T shown in Fig. 10(e) exactly circular.

[0056] In Embodiment 1, the balance in weight of the crank webs is made around 50% so that the track of rotation would be a substantially exact circle. In the couple balancer 6 having the balance vector t, generated is inertia force in a direction shown by broken lines br1 and br2 in Fig. 9. That is to say, forming the couple balancer 6 so that the inertia force would be br1 and br2 allows the primary couple of the engine 1 to be balanced by means of the couple balancer 6, and thereby, to be eliminated.

[0057] Now, described will be a method of determining the balance in weight of the crank webs so that the track of rotation would be a circle. When Mr denotes reciprocation mass of the respective cylinders (unit is a moment g·mm), Mo denotes rotation mass of the respective cylinders (unit is a moment g·mm) and M1 to M8 denote unbalance quantities of two half crank webs for the respective cylinders in order from the first cylinder (a moment g·mm), the balance ratios k of the respective half crank webs are expressed by the following formula:

$$k\ (1 \text{ to } 8) = \{M(1 \text{ to } 8) - M_0/4\}/Mr$$
$$= \{M(1 \text{ to } 8) - W_{rot} \cdot r/4\}/W_{rec} \cdot r$$

The unbalance quantities M1 to M8 are obtained by a

formula of: M1 to M8 = 0.25 x Mr + M_0 /4, for example, for the purpose of making the compound vector of the primary inertia force in all of the cylinders constant (circular). In the case that M1 is same as M4 and M2 is same as M3, the balance ratio k of one of a pair of half crank webs can be set at less than 0.25 while the balance ratio k of the other can be set to be large by the quantity corresponding to that of the above.

[0058] The half crank web away from the center of the crankshaft influences the couple much more than the half crank web close to the center of the crankshaft since a couple is proportional to the distance D (D_L and D_R) from the center of a crankshaft. Accordingly, making the balance ratio (k_L) of the half crank web away from the center of the crankshaft large allows the balance ratio (k_L) of the half crank web close to the center of the crankshaft to be made small, so that the sum of them (k_L + k_R) can be made smaller than 50%.

[0059] In Embodiment 1, the balance ratios k (1 to 8) of the respective half crank webs are set as shown in Fig. 11. That is to say, for the first and fourth cylinders, the balance ratios of the crank webs 21a and 24a located outside the engine are set at 0.427 while the balance ratios of the crank webs 21b and 24b located inside the engine 1 are set at 0.025 so that the total balance ratio of the both crank webs would be 0.452. For the second and third cylinders, the balance ratios of the crank webs 22a and 23a located outside the engine 1 are

set at 0.357 while the balance ratios of the crank webs 22b and 23b located inside the engine 1 are set at 0.017 so that the total balance ratio of the both crank webs would be 0.374.

[0060] It may be possible to set all of the balance ratios k_{1L}, k_{1R}, k_{4R} and k_{1L} of the crank webs 21a, 21b, 24a and 24b for the first and fourth cylinders at 0.25 and the balance ratios k_{2L}, k_{2R}, k_{3R} and k_{3L} for the second and third cylinders at a value other than 0.25 ($k_{2L} = k_{3R} = 0.357$ and $k_{2R} = k_{3L} = 0.017$, for example). Contrary to the above, all of the balance ratios k_{2L}, k_{2R}, k_{3R} and k_{3L} for the second and third cylinders may be set at 0.25 while the balance ratios k_{1L}, k_{1R}, k_{4R} and k_{4L} for the first and fourth cylinders may be set at a value other than 0.25 ($k_{1L} = k_{4R} = 0.427$ and $k_{1R} = k_{4L} = 0.025$, for example).

[0061] The couple balancer 6 is provided at a location away from the crankshaft 3 rearward (rearward along the Y axis shown in Fig. 9) as shown in Figs. 1 to 4 while the first and second balance weights 12 and 13 are located so as to correspond to the crank pins 22 and 23 of the second and third cylinders. The first and second balance weights 12 and 13 in Embodiment 1 are formed in the shape of a fan from a view in the direction of the axial line. The first and second balance weights 12 and 13 are formed so as to be faced to a space between the crank webs 22a and 22b for the second cylinder and a space between the crank webs 23a and 23b for the third cylinder when they

move to a location most close to the crankshaft. In the first and second balance weights 12 and 13, built in is heavy metal 31 for the purpose of adjusting the mass. The first balance weight 12 is formed to generate the inertia force shown by br1 in Fig. 9. The second balance weight 13 is formed to generate the inertia force shown by br2 in Fig. 9.

[0062] The couple balancer 6 in Embodiment 1 is located at a place close enough to the crankshaft 3 so that the first and second balance weights 12 and 13 would not contact with the large end portion 27b of the connecting rod 27, as shown in Figs. 4(a) to 4(e). Fig. 4(a) shows locations of the crank pin 22 of the second cylinder and the first balance weight 12 when the piston 28 of the first cylinder is located at the upper dead point. Fig. 4(b) shows a state in which the crankshaft 3 is rotated by 180° from the location in the state shown in Fig. 4(a). Fig. 4(c) shows a state in which the crankshaft 3 is rotated by 217.5° from the location in the state shown in Fig. 4(a). Fig. 4(d) shows a state in which the crankshaft 3 is rotated by 225° from the location in the state shown in Fig. 4(a). Fig. 4(e) shows a state in which the crankshaft 3 is rotated by 270° from the location in the state shown in Fig. 4(a). The locations of the large end portion 27b of the connecting rod 27 and the second balance weight 13 for the third cylinder are similar to the locations in Fig. 4 other than a point that the phase is different by 180°. Providing the

balance weights for the first and fourth cylinders enables the shape to be advantageous in avoiding contact with the large end portion 27b of the connecting rod. That is to say, it is possible to make the shape of a fan large.

[0063] In the in-line four-cylinder engine 1 having the above-mentioned structure, the primary inertia force, the secondary inertia force and the secondary couple can be eliminated only by means of the crankshaft 3 while the remaining primary couple can be eliminated by means of the couple balancer 6. The couple balancer 6 is not necessary to be provided on a side opposite to the cylinder with respect to the crankshaft 3 and can be provided by the side of the crankshaft 3 (a rear side or a front side of a vehicle body), as exemplified in Embodiment 1.

[0064] Accordingly, providing the couple balancer 6 by the side of the crankshaft 3 allows a compact in-line four-cylinder engine in an axial direction of the cylinder to be provided. In such an engine 1, the couple balancer 6 does not agitate oil in the oil pan 5a, so that loss in power is little and the capacity of the oil pan 5a can be made large.

[0065] Further, in the engine 1 in Embodiment 1, it is arranged that a part of the first balance weight 12 of the couple balancer 6 be faced to a space between a pair of half crank webs 22a and 22b for the second cylinder of the crankshaft 3 while the second balance weight 13 of the couple balancer 6

be faced to a space between a pair of half crank webs 23a and 23b for the third cylinder. Accordingly, the couple balancer 6 can be made compact in the axial direction and provided closely enough to the crankshaft 3.

Therefore, the in-line four-cylinder engine can be made more compact.

[0066] Moreover, the balance weights may be faced to a space between the pair of half crank webs for the first cylinder and a space between the pair of half crank webs for the fourth cylinder. In this case, the weight can be decreased more than the former case, so that a load of a bearing can be reduced. In addition, in the above case, the shape can be advantageous in avoiding contact of the balance weights with the large end part 27b of a connecting rod as described above. This allows a degree of freedom in designing the shape of the balance weight to be increased, so that there is a possibility of achievement of a more compact engine.

[0067] Now, described will be a reason why setting the balance ratios k (1 to 8) of the respective half crank webs as shown in Fig. 11 allows a track of a vector of the primary inertia couple to be formed into a circle as a premise of the invention, made reference to Fig. 13.

[0068] D_1 to D_4 denote distance from the center C of the crankshaft to the center of the respective cylinders while D_{1L}, D_{1R}, D_{2L}, D_{2R}, D_{3L}, D_{3R}, D_{4L} and D_{4R} denote distance to the half

crank webs of the respective cylinders. Further, k_{1L}, k_{1R}, k_{2L}, k_{2R}, k_{3L}, k_{3R}, k_{4L} and k_{4R} denote the balance ratios k (1 to 8) of the respective half crank webs in order from the first cylinder.

[0069] The crankshaft is assumed to be symmetrical with respect to the center thereof for the purpose of simplifying calculations. In this case, the following formulas are satisfied:

$$D_1 = D_4$$
$$D_2 = D_3$$
$$D_{1L} = D_{4R}$$
$$D_{1R} = D_{4L}$$
$$D_{2L} = D_{3R}$$
$$D_{2R} = D_{3L}$$
$$k_{1L} = k_{4R}$$
$$k_{1R} = k_{4L}$$
$$k_{2L} = k_{3R}$$
$$k_{2R} = k_{3L}$$

[0070] The primary inertia couple M(1) is calculated on the basis of the above premise. That is to say, a couple generated at the center C of the crankshaft by the primary inertia force F(1) of each cylinder is obtained for every cylinder and the sum of them is denoted by M(1).

[0071] The sum M(1) can be obtained as described below.

First, the first cylinder is taken into consideration. The X and Y axes are provided as shown in Figs. 5 and 13 wherein

the Y axis is an imaginary number axis. The following formulas can be obtained:

$$M_{1L} = iD_{1L}\{k_{1L} \cdot F \cdot e^{i(\theta+\pi)}\}$$
$$M_{11} = iD_{1} \cdot F \cdot \cos\theta$$
$$M_{1R} = iD_{1R}\{k_{1R} \cdot F \cdot e^{i(\theta+\pi)}\}$$

wherein M_{1L} and M_{1R} denote the moment operating on the left and right half webs 21a and 21b and M_{11} denotes the moment (in an opposite direction to that of M_{1L} and M_{1R}) operating on the center of a cylinder.

This is true of the second to fourth cylinders, except for θ. Accordingly, adding the above to the four cylinders to arrange the formulas allows M(1) to be obtained.

[0072] The following relations (2) and (3) can be used for calculation so as to obtain M(1) by means of the following formula (4):

$$D_{1L}k_{1L} + D_{1R} \cdot k_{1R} = D_1 k_1 \equiv A \quad \cdots (2)$$
$$D_{2L}k_{2L} + D_{2R}k_{2R} = D_2 k_2 \equiv B \quad \cdots (3)$$
$$M(1) = (D_1 k_1 \sin\theta + D_2 k_2 \cos\theta) \cdot 2 \cdot F + \{D_1(1-k_1)\cos\theta + D_2(k_2-1)\sin\theta\}2Fi \quad \cdots (4)$$

In the case of $k_1 = k_2 = 0.5$ (the balance ratio is 50%),

$$M(1) = (D_1 \sin\theta + D_2 \cos\theta)F + (D_1 \cos\theta + D_2 \sin\theta)Fi \quad \cdots (5)$$

[0073] The formula (5) expresses a circle of the radius $F \cdot \{(D_1^2 + D_2^2)\}^{1/2}$. Accordingly, it can be seen that the primary balancer can be used for an offset in the case of $k_1 = k_2 = 0.5$. Now, the first cylinder will be taken into consideration.

pin 21 for the first cylinder is just located at the upper end in the drawing in the X axis (an axial line of the cylinder).

[0048] In the condition, a resultant force of the primary inertia force due to the mass of the reciprocating portion (the piston 28, the piston pin 28a and the small end part 27a of the connecting rod 27) and the primary inertia force due to the mass of the rotating portion (the crank pin 21, the large end part 27b of the connecting rod 27 and the crank webs 21a and 21b) are operated upward in the first cylinder. This is because the crankshaft 3 is formed so that the balance would be about 50%, as descried above, and the inertia force of the rotating portion, which operates downward, becomes about a half of the inertia force of the reciprocating portion, which operates upward. That is to say, in the first cylinder, a compound vector of the primary inertia force due to the reciprocation mass and the rotation mass is directed upward in parallel to the X axis. The size of the compound vector of the first cylinder varies as shown in Fig. 10(a). A direction of rotation of the compound vector is counterclockwise, inversely to the direction of rotation of the crankshaft.

[0049] In the second cylinder, the inertia force of the reciprocating portion is substantially 0 since the piston 28 is located at a substantially middle point on the way from the lower dead point to the upper dead point. Accordingly, only

When a standard value of the balance ratios k_{1L} and k_{1R} is assumed to be 0.25, the deviation from the standard value is (k_{1L} - 0.25). It is known that k_1 = 0.5 when the following formula (6) is satisfied in the above condition.

$$(k_{1L}-0.25)/(0.25-k_{1R})=D_{1R}/D_{2R} \quad \cdots(6)$$

$$k_{1L}=(k_{1R}/k_{1L})(0.25-k_{1R})+0.25$$

$$A=0.25(D_{1R}+D_{1R})$$
$$=0.25\cdot 2D_{1}$$
$$=0.5D_{1}$$

wherein it is meant that A is defined by the formula (2) and k_1 = 0.5 in the formula.

[0074] In the case of the second cylinder, it is also arranged to be k_2 = 0.5 when the following formula is satisfied:

$$(k_{2L}-0.25)/(0.25-k_{2R})=D_{1R}/D_{2R} \quad \cdots(7)$$

Accordingly, the vector M(1) is formed into a circle when the formulas (6) and (7) are satisfied. The balance ratios k_{1L}, …, k_{4L} and k_{4R} in Fig. 11 are determined on the basis of the formulas (6) and (7). It can be seen that $k_{1L} + k_{1R}$ = 0.452 and $k_{2L} + k_{2R}$ = 0.374, the weight of the half crank web is {1 - (0.452 + 0.374)} = 0.174 and the crankshaft can be reduced in weight by around 17% compared with the case of $k_{1L} = k_{1R}$ = 0.25, in the above case.

EMBODIMENT 2

[0075] In Embodiment 1, the crank pins of the crankshaft are in the STD arrangement shown in Fig. 12(A). In this case,

all of the primary inertia force, the secondary inertia force and the secondary inertia couple can be made substantially 0 because of symmetry of the respective cylinders, so that the remaining primary inertia couple is only offset by means of the primary balancer. In the case, the additional weight W_{add} of a crank web of each cylinder is set at 1/2 of the weight of the reciprocating portion (the balance ratio is 50%) and the weight of the crank web is divided between a pair of left and right half crank webs. That is to say, the balance ratio (k_L) of a half crank web whose distance from the center of the crankshaft is large (namely, which is far from the center) is made large (weighted by 25% or more) while the balance ratio (k_R) of a half crank web whose distance from the center of the crankshaft is small (namely, which is near from the center) is made small (lightened by 25% or less) in order to make the total balance ratio ($k_L + k_R$) 50% or less. This enables the crankshaft to be reduced in weight.

[0076] It has been found, however, that the expected effect can be also achieved even when the invention is applied to a crankshaft other than the crankshaft in the STD arrangement. In other words, it has been found that no inconvenience will practically arise although the primary inertia force and the secondary inertia force remain a little and the primary inertia couple M(1) is formed into not an exact circle but a little flat ellipse, in the case of arrangement shown in Figs. 12(B)

and 12(C).

[0077] In Fig. 12(B), the crank pins of the first and third cylinders are provided on the first plane while the crank pins of the second and fourth cylinders are provided on the second plane. Such a form of the crankshaft is herein referred to as the 90-I arrangement. There can be three kinds of the crank form in accordance with combination of the cylinders (mirror type) in which the balance ratios k_L and k_R are symmetrical (mirror-symmetrical). They are referred to as 90-I-1, 90-I-2 and 90-I-3 as shown in Table 1.

[0078] [Table 1]

Crank form and web balance ratio

Crank form \ Web balance ratio	#1		#2		#3		#4		Mirror type
	K_{1L}	K_{1R}	K_{2L}	K_{2R}	K_{3L}	K_{3R}	K_{4L}	K_{4R}	
STD	0.427	0.25	0.357	0.017	0.017	0.357	0.357	0.017	Mirror symmetry between #1 and #4, mirror symmetry between #2 and #3
90-I-1	0.427	0.25	0.357	0.017	0.025	0.017	0.357	0.017	Mirror symmetry between #1 and #3, mirror symmetry between #2 and #4
90-I-2	0.427	0.25	0.357	0.017	0.017	0.017	0.357	0.017	Mirror symmetry between #1 and #4, mirror symmetry between #2 and #3
90-I-3	0.427	0.25	0.025	0.427	0.357	0.427	0.025	0.427	Mirror symmetry between #1 and #2, mirror symmetry between #3 and #4
90-J-1	0.427	0.25	0.357	0.017	0.017	0.017	0.357	0.017	Mirror symmetry between #1 and #4, mirror symmetry between #2 and #3
90-J-2	0.427	0.25	0.025	0.427	0.025	0.427	0.025	0.427	Mirror symmetry between #1 and #2, mirror symmetry between #3 and #4
90-J-3	0.427	0.25	0.357	0.017	0.025	0.017	0.357	0.017	Mirror symmetry between #1 and #3, mirror symmetry between #2 and #4

[0079] In Fig. 12(C), the crank pins of the first and second cylinders are provided on the first plane while the crank

pins of the third and fourth cylinders are provided on the second plane. Such a form is herein referred to as the 90-J arrangement. There can be three kinds of the crank form in accordance with combination of the cylinders (mirror type) in which the balance ratios k_L and k_R are symmetrical (mirror-symmetrical). They are referred to as 90-J-1, 90-J-2 and 90-J-3 as shown in Table 1.

[0080] Table 2 shows the primary inertia force F(1), the secondary inertia force F(2) and the primary inertia couple M(1) and the secondary inertia couple M(2), which are calculated in a case that the balance ratios k_L, k_R, ... k_{4L} and k_{4R} of half crank webs of the respective cylinders are set as shown in Table 1 in the above crank forms. In the calculation, it is assumed that displacement per a cylinder of an engine is equal to 250cc and a rotational speed is constant.

[0081] [Table 2]

Result of calculation of unbalance inertia force

Crank form / Remaining unbalance force	STD	90-I-1	90-I-2	90-I-3	90-J-1	90-J-2	90-J-3
Primary inertia force: F(1) ratio	0	0	a ≠ 0	a ≠ 0	a ≠ 0	0	a ≠ 0
Secondary inertia force: F(2) ratio	0	0	0	0	0	0	0
Primary inertia couple: M(1)	0	0.81 to 0.99 (oval)	0.89 (circle)	73 to 1.06 (oval)	0.45 (circle)	0.39 to 0.51 (oval)	0.33 to 0.56 (oval)
Secondary inertia couple: M(2) kgm	0	±0.36(X) 0 (Y)	±0.36(X) 0 (Y)	±0.36(X) 0 (Y)	±0.72(X) 0 (Y)	±0.72(X) 0 (Y)	±0.72(X) 0 (Y)

[0082] It can be seen from Table 2 that the primary inertia force F(1) does not become 0 but remains by "a" (≠ 0) in the

90-I-2 and other arrangements. The quantity of "a" is little enough not to cause a practical matter. The secondary inertia force F(2) is 0 in all arrangements. The primary inertia couple M(1) is formed into an ellipse in the arrangements other than the STD, 90-I-2 and 90-J-1 arrangements. Further, it can be seen that the secondary inertia couple M(2) remains in the X axis (in a direction of a cylinder axis) in the arrangements other than the STD arrangement.

[0083] Generally, in an engine for a motorbike, the inertia couple within around ± 30 kgm is not a matter on the basis of a rule of thumb of evaluation of bodily sensed vibrations by a passenger. In view of the evaluation standard, it can be considered that the couples M(1) and M(2) shown in Table 2 are small enough in a range of a general rotational speed of an engine, and therefore, there is no problem in practical use.

EMBODIMENT 3

[0084] Fig. 14 is a side view of a motorbike provided with an engine in accordance with the invention. The motorbike is provided in the vicinity of the center of a vehicle body frame 52 with an engine 50.

[0085] The vehicle body frame 52 is substantially in the shape of a hook in which the rear part of the frame 52 is bent downward in a side view. The engine 50 is mounted in a space, which is enclosed by the vehicle body frame 52 and opens

downward and forward. A crankcase 54 of the engine 50 can be divided between upper and lower parts by a dividing surface 54A sharply inclined to the front side. A crankshaft 56, a couple balancer (a primary balancer) 58 and a main transmission shaft 60 are rotatably held on the dividing surface 54A in this order from the lower-front side to the upper-rear side. An output shaft 62, which functions as an auxiliary transmission shaft, is located under the main shaft 60 and rotatably held in a lower half of the crankcase 54.

[0086] The crankshaft 56, the balancer 58, the main shaft 60 and the auxiliary shaft 62 are parallel in a width direction of the vehicle body. A cylinder body including four cylinders and a cylinder head 64 rise so as to bend forward from an upper-front surface of the crankcase 54. It goes without saying that the crankshaft 56 and the balancer 58 herein have structures in accordance with the invention. The engine 50 is different from the engine 1 in Embodiment 1 in a point that the main shaft 60 is provided on the dividing surface 54A of the crankcase while the output shaft 62 is provided under the main shaft 60 (refer to Fig. 1). Accordingly, the center of gravity of the engine 50 is in a high position and the size in the front-rear direction of the engine 50 is small, so that the inertia moment around the vertical axis becomes small. This causes easy inclination of the vehicle body to a direction of a turn in turning left and right, and thereby, improvement

in turning performance. This is convenient for avoiding an obstacle in a bad road.

[0087] 66 denotes a rear arm whose front end is held in the vehicle body frame 52 by means of a pivot shaft 68 located on the rear side of the output shaft 62 so that the rear arm can rotate to freely swing in a vertical direction. A rear wheel 70 is held at the rear end of the rear arm 66. Rotation of the output shaft 66 is transmitted to the rear wheel 70 by means of a chain 72. A lower-rear end of the vehicle body frame 52 and the rear arm 66 are connected by means of a connector 74 in a substantially triangle shape in a side view and a link 76. A cylinder-shaped buffer 78 is provided between the connector 74 and the vehicle body frame 52 to add downward return force to the rear wheel 70.

[0088] 80 denotes a front fork held at the front end of the vehicle body frame 52 so as to be freely rotatable left and right. A front wheel 82 is held at the lower end of the front fork 80. A steering handlebar 84 is fixed to an upper part of the front fork 80. On the upper side of the vehicle body frame 52, provided are an air cleaner 86 and a fuel tank 88 before and behind each other in this order. The air cleaner 86 takes running wind in from the vicinity of the upper part of the front fork 80 to guide the inhaled air to the respective cylinders from a rear surface of the cylinder head 64 through an intake pipe 90. Fuel is blown into the intake pipe 90 from

a fuel injection valve 92. 94 denotes an exhaust pipe extending from the front surface of the cylinder head 64 to the rear side through the front and lower sides of the crankcase 54. 96 denotes a driving seat.

[0089] In accordance with Embodiment 3, the crankshaft 56 is of two-plane type, and therefore, vibrations characteristic in S/N of driving torque, amplitude of vibrations, frequency and the like can be transmitted to the vehicle body frame 52. This enables a driver to feel the driving torque of an engine all over his or her body. Accordingly, the acceleration feeling is improved, and thereby, a driving sense is extremely improved.